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(54) Condenser.

(57) A condenser (10) for use in a car cooling system, comprising a pair of parallel headers (13, 14), a plurality of tubular elements (11) whose opposite ends are connected to the headers (13, 14) and fins (12) provided in the air paths between adjacent tubular elements (11), each of the headers (13, 14) being made of a cylindrical pipe of aluminum and each of the tubular elements (11) being made of a flattened hollow tube of extruded aluminum, the opposite ends of the tubular elements (11) being inserted into slits (15) provided in the headers (13, 14) and soldered therein so as to be liquid-tight.

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The present invention relates to a condenser for use as a cooler in automobiles and more particularly to a condenser for such us which is preferably made of aluminum which includes aluminum alloys.

The present application is divided from copending Application No. 89202415.9.

In general heat exchangers as car coolers use a high pressure gaseous coolant, and they must have an anti-pressure construction.

To this end the known heat exchangers are provided with a core which includes flat tubes arranged in zigzag forms, each tube having pores and fins interposed between one tube and the next. Hereinafter this type of heat exchanger will be referred to as a serpentine type heat exchanger.

The serpentine type heat exchangers are disadvantageous in that the coolant undergoes a relatively large resistance while flowing throughout the tubes. To reduce the resistance the common practic is to use wider tubes so as to increase the cross-sectional area thereof. However this leads to a large core and on the other hand an accommodation space in the automobile is very much limited. As a result this preactice is not always effective.

Another practice is to placing more fins by reducing the distances between the tubes. This requires that the height of each fin is reduced. However, when the fins are too small the bending work becomes difficult, and takes more time and labour.

In general the condenser has a coolant path which consists of two sections, that is, an inlet section, hereinafter referred to as "condensing section" in which the coolant is still gaseous and an outlet section, hereinafter referred to as "supercooling section" in which it becomes liquid. In order to increase the heat exchange efficiency it is ssentail to increase the area for effecting heat transfer in the condensing section, whereas it is no problem for the supercooling section to have a reduced area for heat transfer.

The conventional serpentine type heat exchangers have a coolant passageway which consists of a single tube. It is impossible for a single tub to be large in some part and small in others. If the tube is to have a wider cross-sectional section the tube per se must be large throughout the entire length; in other words a large tube must be used. This of course leads to a larger condenser.

As is evident from the foregoing description it is difficult to improve the conventional serpentine type heat exchangers merely by changing the dimensional factors thereof.

Basically the serpentine type heat exchangers involve the complicated process which consists of bending tubes and then assembling them into a core in combination with fins. This is why it is difficult to produce the heat exchangers on automatic mass production line. Non-automatic production is costly.

Condensers of the type envisaged in this application are detailed in US-A-1 958226 and WO 84/01208.

US-A-1 958226 discloses a condenser which comprises flat hollow tubular elements inserted in slits in the headers and internal partitions in the headers, but the partition is not inserted in the header through a slit.

The present invention aims at solving the difficulties pointed out with respect to the conventional serpentine type heat exchangers and has for its object to provide a condenser having a relatively small core which nevertheless includes a large effective cross-sectional area for coolant passageways, thereby reducing a possible resistance to the flow of coolant.

According to the present invention a condenser for use in an air-conditioning system, comprises a plurality of flat tubular elements disposed in parallel with each other, a plurality of fin members each interposed between adjacent tubular elements and a pair of hollow headers to which both ends of each tubular elements are fluid-tightly connected, is characterized in that each header comprises a header pipe which is made of a brazing sheet composed of a core sheet and a brazing agent layer coated on at least one surface of the core sheet and having side portions extending longitudinally and abutment bonded to each other, wherein the opposite ends of the tubular elements are inserted in slits provided in the header pipes and liquid-tightly brazed therein.

Also according to the invention a process for producing a condenser for use in an air-conditioning system comprising a plurality of flat tubular elements disposed in parallel with each other and a pair of hollow headers to which both ends of each tubular elements are fluid-tightly connected, is characterized in that the process comprising preparing a pair of header pipe which is made of a brazing sheet composed of a core sheet and a brazing agent layer coated on at least one surface of the core sheet, providing slits in the header pipe, and inserting the ends of the tubular elements in the slits so as to fabricate a provisional assembly of the tubular elements and the header pipes, and heating the provisional assembly to effect the permanent joint between the tubular elements and the header pipes.

In general, the multi-flow pattern system is difficult to withstand a high pressure provided by a pressurized gaseous coolant because of the relatively fragile joints between the headers and tubular elements and th headers per s which are constructed without presupposing the high pressure

which would act thereon by the coolant. In order to solve this problem encountered by the multi-flow pattern system the condenser of th present invention uses a cylindrical pipe for the header, and flat tubes for the tubular elements, whose opposite ends are inserted in the slits produced in the headers and soldered therein thereby ensuring that the condenser withstands a high pressure provided by the coolant.

Each of the headers is internally divided by a partition into at least two sections; this is, a condensing section and a supercooling section, wherein the condensing section has a coolant in its gaseous state whereas the supercooling section has a coolant in its liquid state. When the coolant is in a gaseous state its volume is large, which requires a relatively large effective cross-sectional ar a for the coolant passageways. When it is in a liquid state the volume reduces, thereby allowing the coolant passageway to have a relatively small cross-sectional area.

In this condenser the dimensional relationships among the width, height and pitch of the tubular elements and fins are as follows:

Width of the tubular elements:
6 to 12mm
Height of the tubular element:
5mm or less
Height of each fin:
8 to 16mm
Fin Pitch:
1.6 to 3.2mm

The tubular elements are jointed to the headers; more specifically, the opposite ends of each tubular element are inserted into slits produced in the headers so that they fit therein in a liquid-tight manner and then they are soldered therein. Prior to th insertion the tubular elements or the headers or both are provided with a layer of a soldering substance. All the soldering is effected at one time by placing the assembled unit in a furnace, thereby saving time and labour in the assembling work.

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:-

Fig. 1 is a front view showing a condenser embodying the present invention;
Fig. 2 is a plan view showing the condenser of Fig. 1;
Fig. 3 is a perspective view showing the joint between the header and the individual tubes;
Fig. 4 is a cross-sectional view through the line 4-4 in Fig. 1;
Fig. 5 is a cross-sectional view showing th joint between the header and the tube;
Fig. 6 is a cross-sectional view of the tube ex mplifying a dimensional relationship about it;
Fig. 7 is a cross-sectional view of th fin xemplifying a dimensional relationship about it;
Fig. 8 is an explanatory view showing a flow pattern of coolant;
Fig. 9 is a perspective view showing a modified version of the joint between the tubes and the header;
Fig. 10 is a cross-sectional view showing the relationship between the tube and the header after they are jointed to each other;
Fig. 11 is a cross-sectional view showing a modified version of the stopper produced in the tube;
Fig. 12 is a cross-sectional view showing another modified version of the stopper;
Fig. 13 is a cross-sectional view showing a further modified version of the stopper;
Fig. 14 is a graph showing the relationship between the width of the tubes and the rate of air passage therethrough;
Fig. 15 is a graph showing the relationship between the height of the tubes and the pressure loss of air; and
Fig. 16 is a graph showing variations in the heat exchange efficiency with respect to the height of the fins and the pressure loss of air.

As shown in Fig. 1 the condenser 10 of the present invention includes a plurlaity of planar tubes 11, and corrugated fins 12 alternately arranged. The tubes 11 are connected to headers 13 and 14 at their opposite ends.

The tube 11 is planer, made of aluminum; preferably, of a multi-hollow type.

The header 13, 14 is made of a cylindrical pipe of aluminum. It is provided with slits 15 produced at equal intervals along its length, where the ends of the tubes 11 are soldered to the respective headers 13, 14. The left-hand header 13 is provided with a coolant inlet pipe 16 at its upper end and a plug 17 at the lower end. The right-hand header 14 is provided with a coolant outlet pipe 18 at its lower end and a plug 19 at its upper end. The coolant inlet and outlet are diametrically located. The reference numerals 23 and 24 denote side plates fixed to the fins 12 located at the outermost positions.

Each header 13, 14 is provided with a partition 20, 21, respectively, thereby dividing the internal chamber into upper and lower sections, wherein the partition 20 in the header 13 is located slightly toward the inlet 16, whereas the partition 21 in th header 14 is located aobut 1/3 the length toward the outlet 18.

Because of the provision of the partitions 20 and 21 in th headers 13 and 14 the flow pattern of th coolant is formed as shown in Fig. 8; that is, the coolant passageway is grouped into an inlet section (A), a middle section (B) and an outlet section (C). As seen from Fig. 8 the coolant flows

in thre different directions. In addition, the tubes are different in number from group to group; that is, the group (B) has more tubes than the group (C) (outlet section), and the group (A) (inlet section) has more tubes than the group (B). This means that the group (A) has a larger effective cross-sectional area for coolant passageway than the group (B), which in turn has a greater area for it than the group (C).

Referring to Fig. 8 the coolant introduced into the core through the inlet pipe 16 flows to the right-hand header 14 in the inlet section (A), and then in a reversed direction in the middle section (B). In the outlet section (C) the flow of coolant is again reversed and led to the right-hand header 14, where it is discharged through the outlet pipe 18. While the coolant is flowing through the sections (A), (B) and (C) heat exchange takes place between th coolant and the air passing through the fins 12. In the inlet section (A) the coolant is in its gaseous state, but because of the large effective cross-sectional area in the section (A) heat exchange proceeds efficiently between the coolant and the air. In the section (C) the coolant is in its liquid state, and reduced in its volume, which allows the section (C) to have a relatively small cross-sectional area for coolant passageway as compared with the section (B). In this way the coolant passes through the first condensing section (A), the second section (B) and the third supercooling section (C), in the course of which heat exchange smoothly and efficiently takes place.

In the illustrated embodiment the numbers of tubes are progressively decreased from the section (A) to the sction (B) and to the section (C). However it is possible to give the same number of tubes to the sections (A) and (B), and a smaller number of tubes to the section (C). Alternatively it is possible to arrange so that each section (A) to (C) has the same number of tubes but their cross-s ctional areas are progressively reduced from the section (A) to the section (B) and to the section (C). As a further modification the intermediate section (B) can be omitted; in this case the flow pattern is called a two-path system. In contrast, the abovementioned embodiment is called a three-path system. As a still further modification one or more intermediate sections can be added.

The illustrated embodiment has the headers located at the left-hand side and the right-hand side but they can be located at the upper side and the lower side wherein the tubes and fins are vertically arranged.

To joint th tubes 11 to th headers 13, 14 the tubes or the headers or both are previously provided with a layer of a soldering substance on their ajoining surfac s. Mor specifically, as shown in Fig. 3 ther is an aluminum pipe 13a, such as a clad metal pipe, which is used as the headers 13 and 14. Th clad pipe 13a has a layer of a soldering substance 13b. The pipe 13b is electrically seamed but can be made by extrusion or any other known method. For the soldering substance an Al.Si alloy preferably containing 6 to 13% by weight of Si is used. The tubes 11 are inserted in the slits 15 for their end portions to be held therein. Then they are heated together to melt the soldering substance. In this case, as clearly shown in Fig. 5 the ajoining parts of the tube 11 and the clad pipe 13a have fillets 29, whereby the header 13, 14 and the tubes 11 are jointed to each other without gaps interposed therebetween.

Likewise, the corrugated fins 12 can be provided with a layer of a soldering substance, thereby effecting the soldering joint between the fins 12 and the tubes 11 simultaneously when the tubes 11 are jointed to the headers 13, 14. This facilitates the soldering joint among the headers 13, 14, the tubes 11 and the fins 12, thereby saving labour and time in the assembling work. The layer of a soldering substance can be provided in the inner surface of the clad pipe 13a but the place is not limited to it.

The partitions 20, 21 are jointed to the respective headers 13, 14 in the following manner:

The clad pipe 13a is previously provided with a semi-circular slit 28 in its wall, wherein the slit 28 covers half the circumference of the pipe 13a. The partition 20, 21 is made of a disc-shaped plate having a smaller circular portion 20a and a larger circular portion 20b, wherein the smaller circular portion 20a has a diameter equal to the inside diameter of the pipe 13a, and wherein the larger circular portion 20b has a diameter equal to the outside diameter of the pipe 13a. The larger diameter portion 20b is inserted and soldered in the slit 28. The headers 13, 14 and the partitions 20, 21 are preferably provided with layers of soldering substances as described above, so that the soldering joint between them can be performed simultaneously when the tubes 11 are soldered to the headers 13, 14. This finishes the soldering joint among the headers, the tubes, the fins and the partitions at one time. The larger diameter portion 20b fits in the slit 28 so that no leakage of coolant is likely to occur, and that the appearance of an outer surface of the pipe 13a is maintained. In addition, the larger diameter portion 20b is embedded in the slit 28, thereby preventing the partition 20, 21 from being displaced by an unexpected force acting thereon.

As is generally known in the art, a possible pressure loss of air largely depends on the r lativ positional relationship between the tubes 11 and th fins 12. A reduced pressur loss leads to the incr ased heat xchang efficiency. Accordingly, the heat exchang efficiency depends on this posi-

tional relationship between them.

Now, referring to Figs. 7 and 8 this positional relationship will be described:

It is prescribed so that the tub 11 has a width (W) of 6 to 12mm, and a height (Ht) of not smaller than 5mm, and that the fin 12 has a height (Hf) of 8 to 16mm and a fin pitch (Fp) of 1.6 to 3.2mm. Referring to Figs. 14, 15 and 16 the reasons for the presciptions are as follows:

As shown in Fig. 14, if the tube 11 has a width of smaller than 6mm the fin 12 will be accordingly narrower, thereby reducing the number of louvers 12a. The reduced number of louvers 12a leads to less efficient heat exchange. If the tube is wide enough to allow an adequate number of louvers 12a to be provided on the fins 12, the heat exchange efficiency will be enhanced. However if the width (W) of the tube is more than 12mm, the fins 12 will be accordingly widened, thereby increasing its weight. In addition too wide fins and too many louvers are likely to increase resistance to the air passing therethrough, thereby causing a greater pressure loss of air.

If the fins 12 have a height (Hf) of more than 5mm the pressure loss of air will increase. The inside height (Hp) of the tube 11 is preferably not smaller than 8mm. The inside height (Hp) is important in that it defines the size of an effective coolant passageway. If it is smaller than 8mm the pressure loss of coolant will increase, thereby reducing the heat exchange efficiency. In order to maintain a height (Hp) of at least 1.8mm for coolant passageway, the height (Ht) of the tube 11 will have to be at least 2.5mm, inclusive of the thickness of the tube wall.

As shown in Fig. 16, if the height (Hf) of the fin 12 is not larger than 8mm the pressure loss of air will increase, but if it is larger than 16mm the number of fins will have to be reduced, thereby reducing the heat exchange efficiency.

If the pitch (Fp) of fins 12 is smaller than 1.6mm there will occur an interference between the adjacent louvers 12a, thereby amplifying the pressure loss of air. However if it exceeds 3.2mm the heat exchange efficiency will decrease.

Referring to Figs. 9 and 10 a modified version will be described:

This embodiment is characteristic in that it is provided with shoulders 25 which work as stop means to prevent the tube from being inserted to deeply into the header 13, 14. More specifically, the tube 11 includes a body 111 and a head 111a which has shoulders 25 therebetween. The shoulders 25 are adapted to come into abutment with the heater 13, 14 when the tube 11 is insert d into the slit 15.

As modified v rsions of th stop means various xamples are shown in Figs. 11 to 13:

Fig. 11 shows th process of forming stop means 125. In (a) th tub 211 has sharp or acute corners. The corners are cut away in such a manner as to form bulged portions 125, which provid stop means. Fig. 12 shows a tube 311 having round corners, which are split lengthwise in such a manner as to form shoulders 225. Fig. 13 shows a tube 411 having a relatively thin wall, In this case the cutting and splitting are jointly used in such a manner as to form shoulders 325.

Claims

1. A condenser for use in an air-conditioning system, comprising a plurality of flat tubular elements disposed in parallel with each other, a plurality of fin members each interposed between adjacent tubular elements and a pair of hollow headers to which both ends of each tubular elements are fluid-tightly connected, characterized in that each header comprises a header pipe which is made of a brazing sheet composed of a core sheet and a brazing agent layer coated on at least one surface of the core sheet and having side portions extending longitudinally and abutment-bonded to each other, wherein the opposite ends of the tubular elements are inserted in slits provided in the header pipes and liquid-tightly brazed therein.

2. A condenser according to claim 1, characterized in that each header comprises a partition which divides the interior thereof into chambers arranged in series in the longitudinal direction of the header.

3. A condenser according to claim 2, characterized in that a partition is brazed to the header.

4. A condenser according to claim 1, characterized in that longitudinally extending side portions of each header pipe are electrically seamed to each other.

5. A condenser according to calim 1, characterized in that longitudinally extending side portions of each header pipe are brazed to each other.

6. A condenser according to claim 1, characterized in that each header comprises a cap closing at least one end of each header pipe.

7. A condenser according to claim 1, characterized in that each of the flat tubular elements has at l ast on reinforcing wall which connects an upp r wall of the tubular element to a lower wall thereof.

8. A process for producing a condenser for use in an air-conditioning system comprising a plurality of flat tubular elements disposed in parallel with each other and a pair of hollow h aders to which both ends of each tubular elements are fluid-tightly connected, characterized in that the process comprises preparing a pair of header pipes made of a brazing sheet composed of a core sheet and a brazing agent layer coated on at least one surface of the core sheet, providing slits in the header pipe, inserting the ends of the tubular elements in the slits so as to fabricate a provisional assembly of the tubular elements and the header pipes and heating the provisional assembly to ffect the permanent joint between the tubular lements and the header pipes.

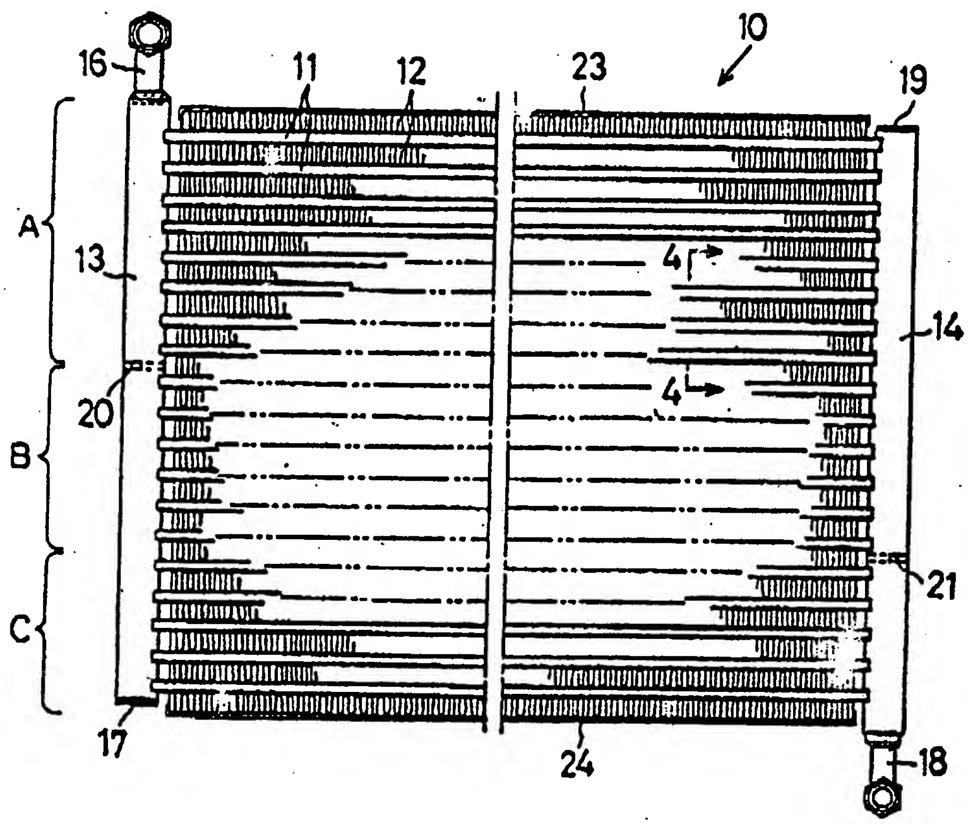

FIG. 1

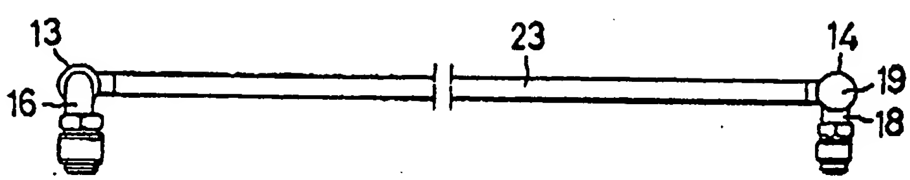

FIG. 2

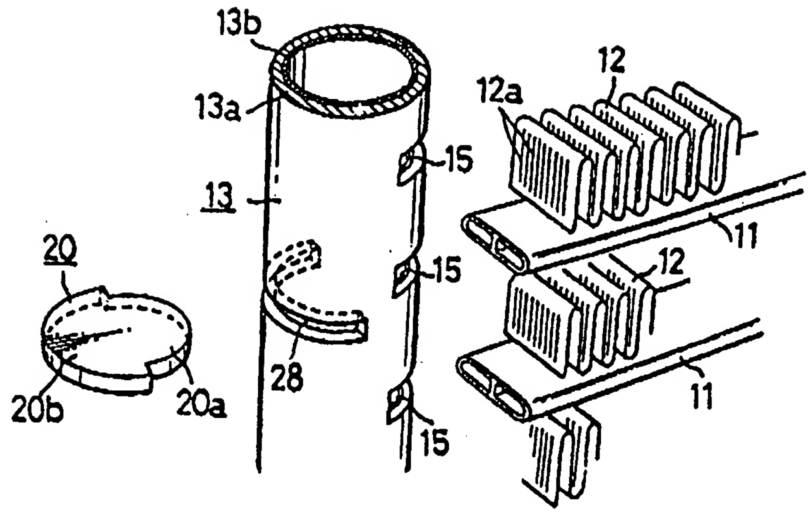

FIG. 3

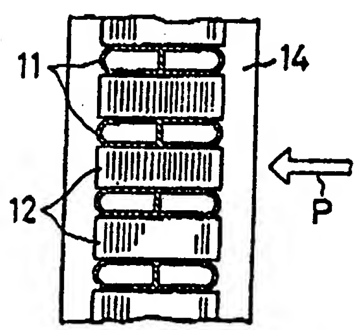

FIG. 4

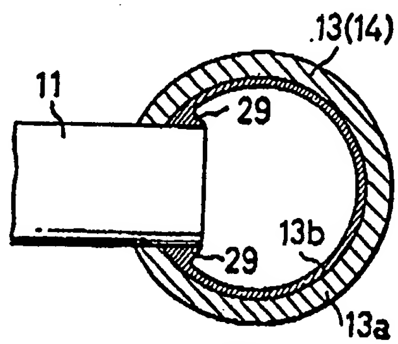

FIG. 5

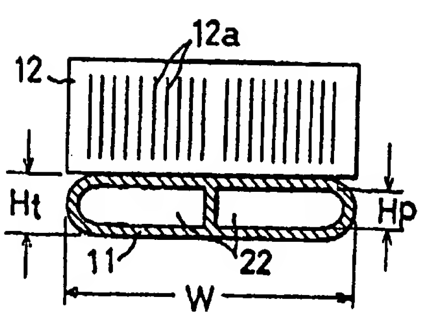

FIG. 6

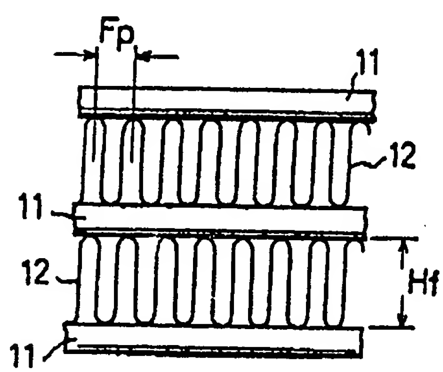

FIG. 7

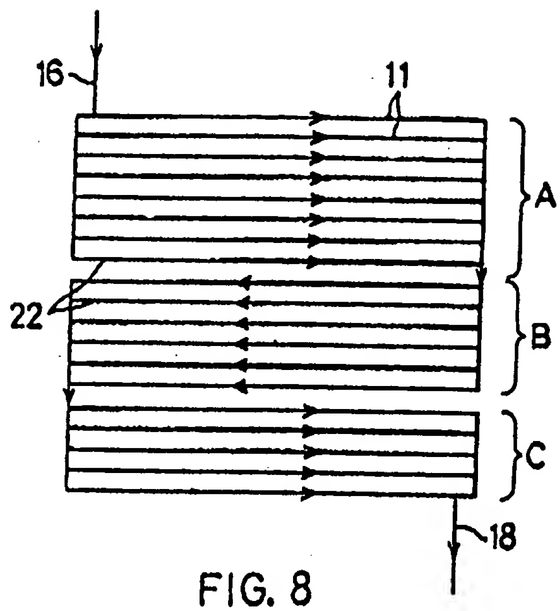

FIG. 8

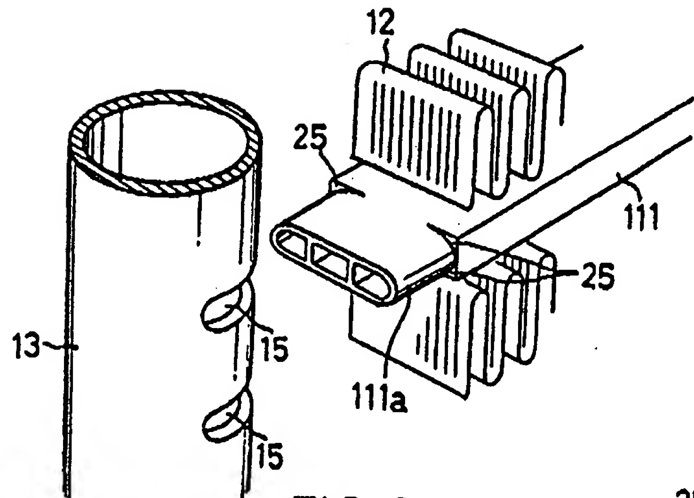

FIG. 9

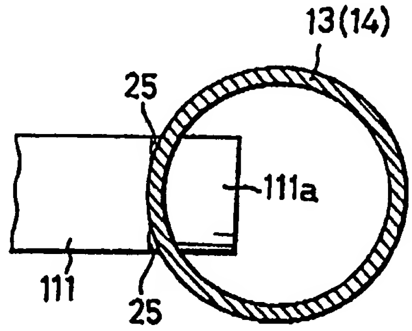

FIG. 10

A8

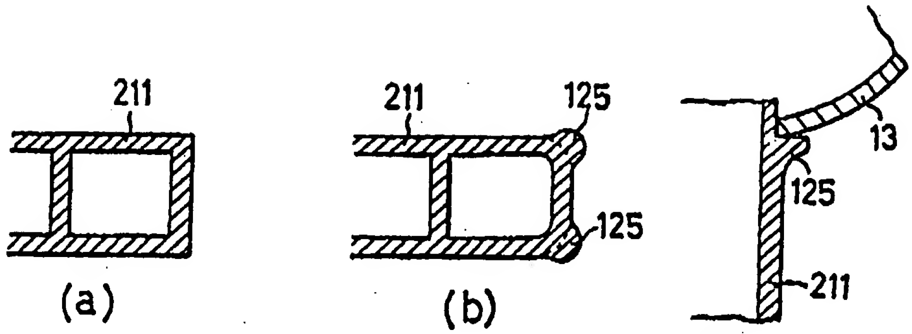

FIG. 11

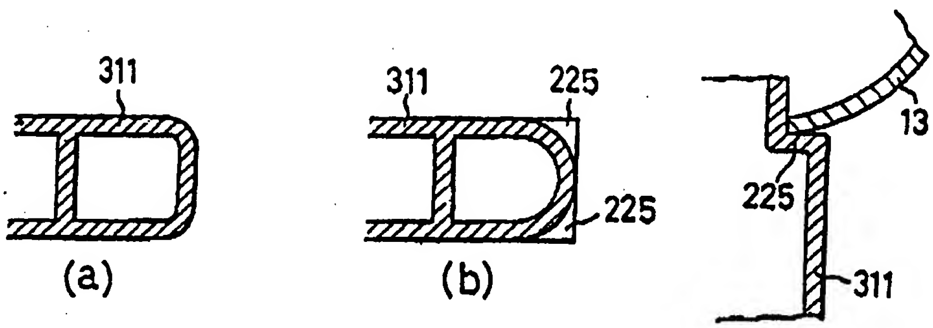

FIG. 12

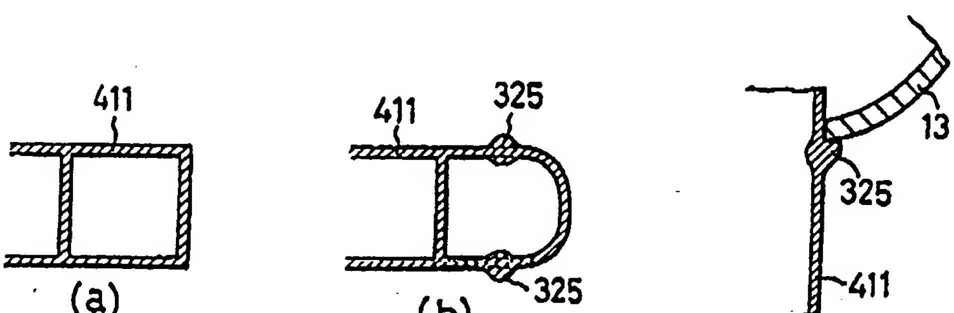

FIG. 13

FIG. 14

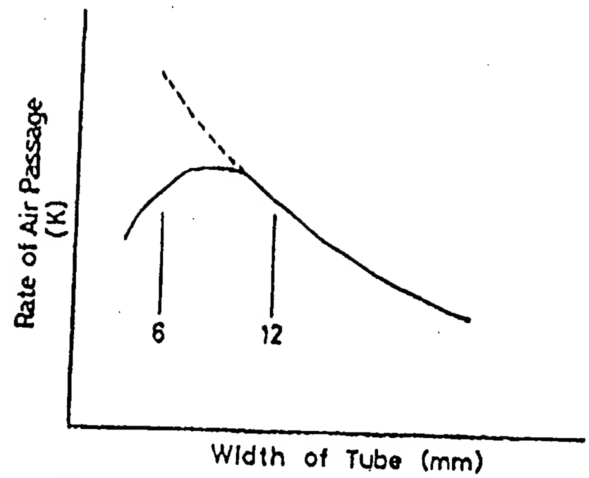

FIG. 15

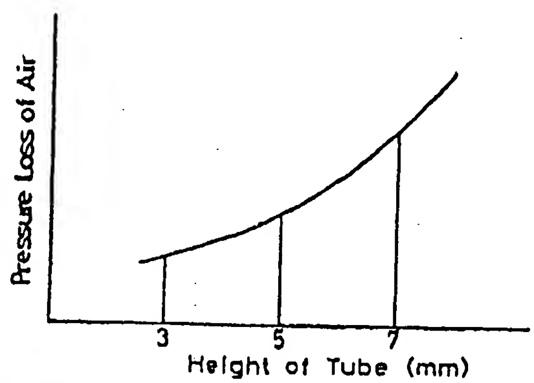

FIG. 16